NASA TECHNICAL MEMORANDUM

NASA TM X-71777

NASA TM X-71777

(NASA-TM-X-71777) EXPERIMENTAL VERIFICATION OF FILM-COOLING CONCEPTS ON A TURBINE VANE (NASA) 14 p HC $3.25 CSCL 21E

N75-29107

Unclas

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TECHNICAL PAPER to be presented at
Winter Annual Meeting of the
American Society of Mechanical Engineers
Houston, Texas, November 30 - December 4, 1975

E-8435

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ABSTRACT

An investigation was conducted to verify some film cooling concepts applied to gas turbine vanes. The film cooling air was ejected from a single row of holes on the convex surface and a double row of holes of the concave surface. Tests were conducted at a gas temperature of 1260 K, a gas pressure of 3 atmospheres, and a coolant temperature of 280 K. Mass-velocity ratios were varied between 0 and 2.0. Data were taken without film-cooling holes, with film-cooling holes but without blowing, and with blowing. A small amount of blowing into a nonturbulent boundary layer caused an increase in vane temperatures. Film cooling when combined with convection cooling was verified to be more effective than either film or convection cooling alone.

NOMENCLATURE

D	film-cooling hole diameter
M	mass-velocity ratio, $(\rho V)_c/(\rho V)_g$
T^*	dimensionless temperature ratio, $(T_w - T_c)/(T_g - T_c)$
V	velocity
w	flow rate
x	surface distance from film-cooling holes
ρ	density

Subscripts:

c	coolant
fc	film coolant
g	mainstream
mc	midchord coolant
tot	sum of convection and film coolant
w	vane wall

INTRODUCTION

Convection cooling of engine components is difficult, and, in some instances, impossible even for the most effective cooling configurations. An analytical comparison of various film and convection cooling schemes is discussed in reference 1. These comparisons show that a combination of convection and film cooling results in a considerable reduction in wall temperatures relative to convection or film cooling alone. In references 2 to 4, the addition of film cooling to the suction (convex) surface of a turbine vane, under certain conditions, results in increased wall temperatures at downstream locations. These references conclude that a nonturbulent boundary layer is tripped by the film-cooling holes and blowing. The boundary layer on the pressure (concave) surface of a turbine vane is usually turbulent over the entire surface downstream of the leading edge region. Consequently, the large increases in Stanton number and skin friction coefficient encountered when a boundary layer is tripped turbulent should not occur with film ejection on the pressure surface. However, blowing does increase the turbulence in the boundary layer.

To verify these concepts, an experimental investigation was performed on a turbine vane in a hot gas environment. Particular attention was given to the turbulence effects caused by a small amount of air ejection. Also considered was the relative effectiveness of film cooling combined with convection as compared with film cooling or convection cooling alone.

The experiment was performed on a J-75 size turbine vane in a four-vane cascade and at gas conditions which simulated a cruise Mach number of 2.7. Coolant flow rates to film-cooling holes in the leading edge region and to convection-cooling passages in the midchord region were individually measured. Vane wall temperatures corresponding to these flow rates were measured (a) without film-cooling holes, (b) with film-cooling holes but without blowing, and (c) with blowing. Calculated momentum thickness Reynolds numbers in the region of the film-cooling holes indicated the existence of a nonturbulent boundary

layer on the suction surface and a turbulent boundary layer on the pressure surface. The film-cooling holes were located near the leading edge where the ratio of momentum thickness to hole diameter was 0.4 on the suction surface and 0.11 on the pressure surface.

APPARATUS

The cascade test facility was designed to operate at average inlet gas temperatures as high as 1645 K and pressures up to 10 atmospheres for the evaluation of various turbine vane cooling configurations. The facility consists of five basic components: (1) an inlet section, (2) a combustor section, (3) a transition section, (4) a test section, and (5) an exhaust section.

The test section was a 23° annular sector of a vane row and contained four vanes and five flow channels. A cross-sectional view of the cascade facility is shown in figure 1(a) while a plan view of the test section is shown in figure 1(b). The central flow channel was formed by the suction surface of vane 2 and the pressure surface of vane 3. The two outer vanes completed the flow channels for the two central vanes and also served as radiation shields between these vanes and the water-cooled cascade walls. Each of the test vanes (vanes 2 and 3) had two separate cooling-air systems, one for the film-cooling flow, and the second for the midchord convection flow. The cooling-air flow rates were metered by turbine-type flowmeters. A more detailed description of this facility is given in reference 5.

Vane Description

A J-75 size vane which had span of 9.78 centimeters and a chord of 6.28 centimeters was used in this investigation. The vane material was MAR M 302. The internal cooling configuration consisted of an impingement-cooled midchord region and a pin-fin-augmented convection-cooled trailing-edge region. This vane was designed to have an impingement-cooled leading edge. However, for this investigation, the leading-edge impingement tube was removed, and the chamber was blocked from the tip end. Therefore, the cooling air entering the vane from

the tip plenum was restricted to cooling the midchord and trailing edge region only. The leading edge chamber served as a plenum for the film-cooling holes which were added to the vane after the initial series of tests. The suction surface of vane 2 had a single row of film-cooling holes while the pressure surface of vane 3 had two staggered rows of film-cooling holes. A composite schematic of the internal cooling scheme is shown in figure 2(a).

The suction surface film-cooling holes were 0.064 centimeter in diameter and equally spaced 2.5 diameters apart. The holes were located 2.11 centimeters downstream of the leading edge stagnation point. They were angled at 28° to the vane surface.

The film-cooling holes in the pressure surface were 0.071 centimeter in diameter and equally spaced 2.2 diameters apart spanwise. The two rows were 1.60 and 1.85 centimeters from the leading edge stagnation point. Both rows were angled at 40° to the vane surface.

INSTRUMENTATION

Eight Chromel-Alumel sheathed thermocouples were located on the suction surface of vane 2 and seven Chromel-Alumel sheathed thermocouples were located on the pressure surface of vane 3 such that these thermocouple stations were adjacent to the central flow channel. A composite sketch of vane 2 and vane 3 thermocouple locations is shown in figure 2(a). The thermocouple locations are shown on the figure with the dimensionless surface distance from the film-cooling holes. The construction and installation of the thermocouples is discussed in reference 6. The cooling-air temperature and pressure were measured at the inlet to the vanes. The mainstream inlet conditions were measured by radially traversing probes.

Each test vane had two separate cooling-air systems which were limited to ambient temperatures for this investigation. The film-cooling air flow and the midchord-convection air flow to each vane were metered individually by turbine type flow meters.

TEST PROCEDURE

The average mainstream inlet total temperature and pressure investigated were 1260 K and 3 atmospheres, respectively. The midspan exit Mach number was 0.85. These gas conditions were scaled from a cruise Mach number of 2.7 and resulted in a nonturbulent boundary layer on the suction surface and a turbulent boundary layer on the pressure surface. Ambient temperature cooling air was used for both the midchord convection-cooling air flow and the film-cooling air flow.

The first series of tests were made without film-cooling holes on the vane. The midchord coolant to mainstream flow ratio w_{mc}/w_g for these tests were varied from 0.0 to 5.6 percent.

A single row of film-cooling holes on the suction surface of vane 2 and a double row of film-cooling holes on the pressure surface of vane 3 were provided for the second series of tests. Film cooling data were then taken at selected values of w_{mc}/w_g. The mass velocity ratio was varied from 0.0 to 2.0.

RESULTS AND DISCUSSION

The analysis of reference 1 shows that a combination of film and convection cooling is an efficient method of maintaining reasonable wall temperatures in some gas turbine applications. However, in certain situations, film-cooling-air ejection can result in higher vane temperatures in contrast to the expected temperature reduction. The data of this paper verify this limitation of film cooling. Additional data are presented which illustrate the benefits of combining film and convection cooling on a turbine vane.

Film Cooling in a Nonturbulent Boundary Layer

Calculated momentum thickness Reynolds numbers indicate the boundary layer on the suction surface of the turbine vane to be in the laminar and transitional regime for the mainstream conditions investigated. Transition from laminar flow was assumed to occur at a momentum-thickness Reynolds number of 200

(ref. 7). Transition to turbulent flow was assumed to occur at a momentum-thickness Reynolds number of 360 (ref. 8). The effects of blowing into a nonturbulent boundary layer are shown in figure 3. Dimensionless vane temperatures are plotted as a function of the dimensionless surface distance from the point of air ejection in figure 3(a). Included in the figure are data for the vane without film-cooling holes, with film-cooling holes but without blowing, and with blowing. Comparing the data for the vanes with and without film-cooling holes, the vane temperatures are shown to increase downstream of the film-cooling holes. The temperature increase in the diffusion region ($x/D > 40$) is shown to be quite substantial. The turbulence created in the boundary layer causes a transition from the nonturbulent to the turbulent regime with a substantial increase in the Stanton number.

A further increase in the vane temperature at locations beyond 15 hole diameters downstream of the film-cooling holes is caused by the injection of a small amount of cooling air into the boundary layer. Note the temperature increase with a mass-velocity ratio increase from 0.0 to 0.15. The addition of blowing causes a further increase in the Stanton number; this increase is due to the increase in the turbulence level of the boundary layer resulting from the addition of a small amount of blowing. A value of the mass-velocity ratio of 0.15 is unrealistically low for a single row of film-cooling holes; however, it represents the value at which the vane temperature reaches a maximum. Beyond a value of 0.15, increases in the mass-velocity ratio are accompanied by decreases in vane temperature.

When the mass-velocity ratio is increased to 1.10, the vane surface is effectively cooled by the film up to an x/D of about 40. The average wall temperature for this mass-velocity ratio is 25 K lower than for the convection cooled vane.

These data are also shown in figure 3(b) as a function of the mass-velocity ratio. The detrimental effect of a small amount of blowing is also shown in this figure for x/D values of 14 and 27 ($0.0 < M < 0.4$). The first location downstream

of the injection point (x/D of 3) is effectively cooled at all mass-velocity ratios.

The data of figure 3(b) show that the wall temperatures decreased with increasing blowing until a minimum was reached. Higher mass-velocity ratios then resulted in increased wall temperatures. The data of Eriksen (ref. 9) indicates that, for a zero pressure gradient, the optimum mass-velocity ratio is between 0.4 and 0.5. Inspection of figure 3(b) shows the optimum for these data to be between 1.0 and 1.5. The momentum thickness to hole diameter is 0.4 and the pressure gradient over the first three thermocouple locations is 2.5 N/cm^2 per cm. These results are consistent with those of Liess (ref. 10) which show an optimum mass-velocity ratio of about 1.1 in a similar pressure gradient, for similar hole ejection angles, and similar momentum thickness to hole diameter ratios.

Film Cooling in a Turbulent Boundary Layer

Favorable film cooling effects. - Calculated momentum thickness Reynolds numbers on the pressure surface of the turbine vane indicate the boundary layer to be turbulent for the mainstream conditions investigated. Figure 4 shows the results of film-cooling air ejection from two staggered rows of holes located in a turbulent boundary layer. Dimensionless vane temperatures are plotted as a function of the dimensionless distance from the film-cooling holes. Vane temperatures are shown to decrease continuously with an increase in the mass-velocity ratio. The pressure surface of the vane reaches a nearly uniform temperature at a mass-velocity ratio of about 0.6. These results support the conclusions of the previous section where the large temperature increases were attributed to a boundary layer transition and an increase in the Stanton number with the addition of film cooling. No optimum value of mass-velocity ratio is evident even though these data were for a pressure gradient of 0.9 $N/cm^2/cm$ and a momentum thickness to hole diameter ratio of 0.11 which are similar to those of Liess. It is possible that an optimum value might exist but for values of mass-velocity ratio greater than 0.9.

Benefits of combining convection with film cooling. - A combination of film and convection cooling results in lower wall temperatures than convection cooling alone. This is shown experimentally in figure 4 and analytically in reference 1. To examine this behavior in more detail, dimensionless temperature distributions are presented in figure 5, for the vane shown in figure 2, with different combinations of coolant flow to show (1) convection cooling only in the midchord region, (2) convection cooling in the midchord plus film cooling near the leading edge, and (3) film cooling only near the leading edge.

Figure 5(a) shows the combination of film and convection cooling to be superior to that of convection cooling alone. Immediately downstream of the double row of holes, the insulating effect of the cool film layer plus the internal cooling from convection results in a low, relatively uniform temperature as indicated by the curve with square symbols. Although the insulating effect of the film decreases further downstream, the vane maintains substantially lower temperatures than the vane cooled by convection only (curve with round symbols). For the test conditions investigated, the average wall temperature of the film-convection cooled vane is 155 K lower than for the convection-cooled vane.

Data for a film-cooled and a film plus convection cooled vane are shown in figure 5(b) for mass-velocity ratios of 1.10 and 0.35, respectively. Here again, the combination of film and convection cooling is superior to that of film cooling alone. With film cooling only in the leading edge region (curve with round symbols) low temperatures are obtained immediately downstream of the film-cooling holes, but the film effectiveness dissipates rapidly and the wall temperature climbs further downstream. The combination of film and convection cooling (curve with square symbols) results in a nearly uniform temperature. The average wall temperature of the film-convection cooled vane is about 45 K lower than the average temperature of the film-cooled vane for the test conditions investigated.

CONCLUSIONS

The experimental investigation of film cooling, as applied to a turbine vane, resulted in the following verifications concerning its limitations and uses:

1. The turbulence created by the ejection of cooling air in regions with other than fully turbulent boundary layers can result in increased wall temperatures downstream of the ejection point.

2. An optimum mass-velocity ratio of between 1.0 and 1.5 was observed on the suction surface in a favorable pressure gradient of 2.5 $N/cm^2/cm$. The pressure surface, with a favorable pressure gradient of 0.9 $N/cm^2/cm$, did not reach an optimum mass-velocity ratio over the range of data investigated ($0.0 < M < 0.9$).

3. Film cooling, when combined with convection cooling, for a given quantity of coolant, was observed to be more beneficial than using the same quantity of coolant for either convection or film cooling alone.

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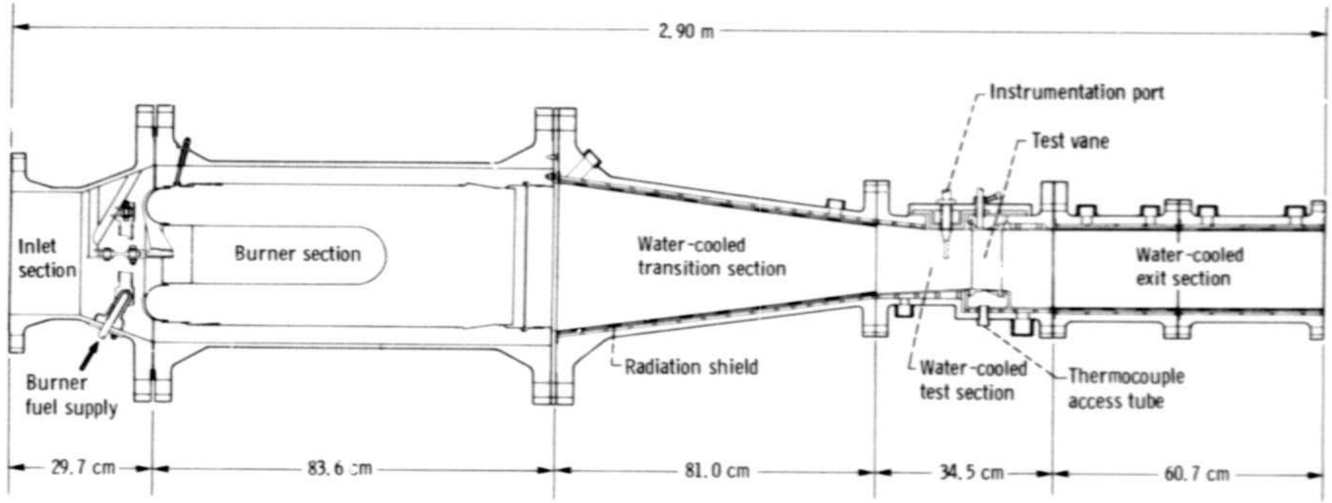
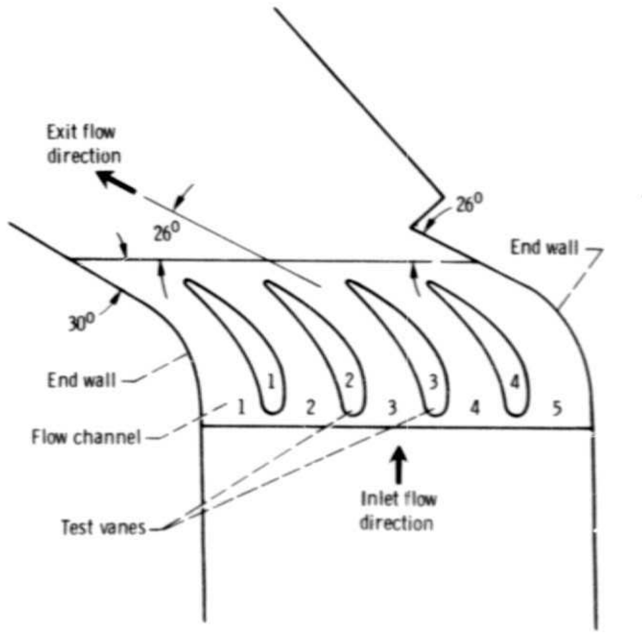
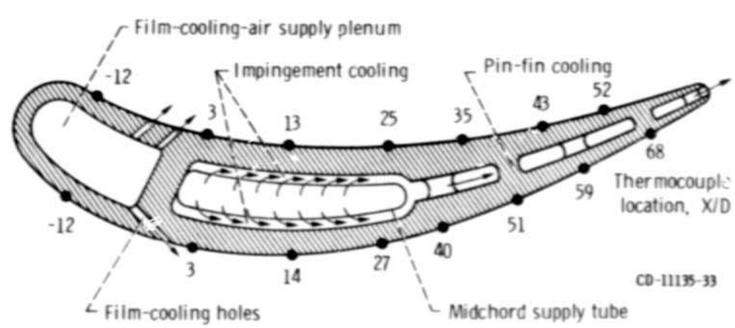


(a) Cascade facility.

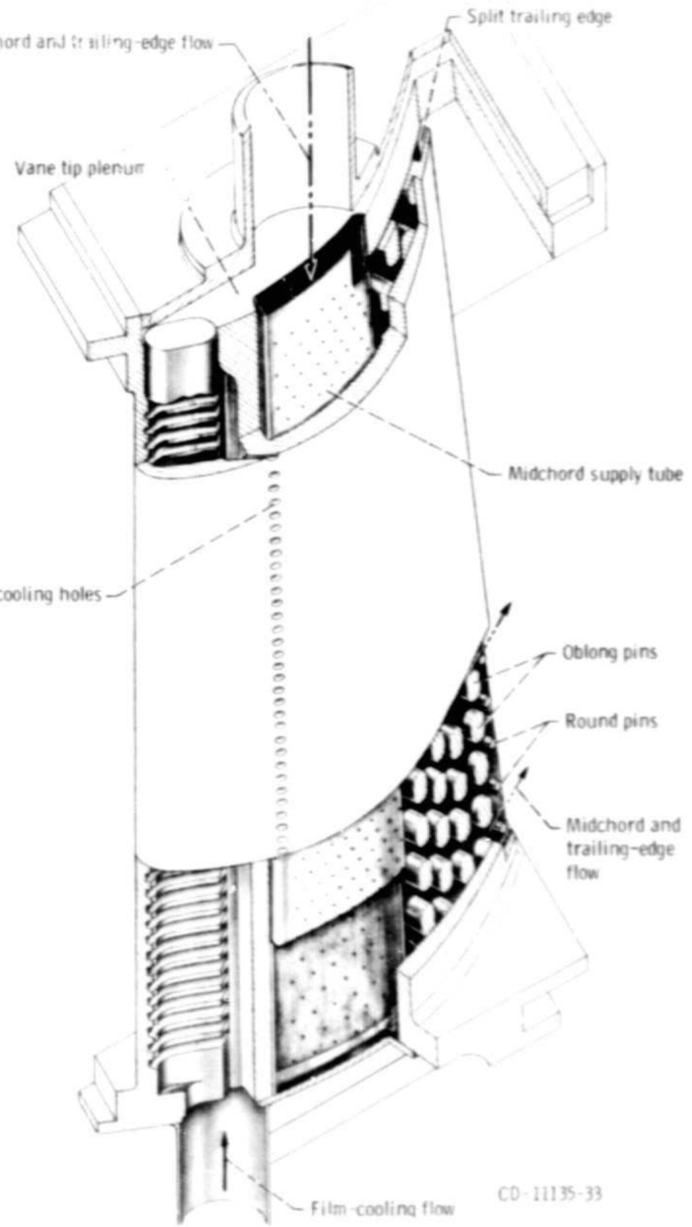


(b) Plan view of cascade test section.

Figure 1. - Schematics of experimental apparatus.

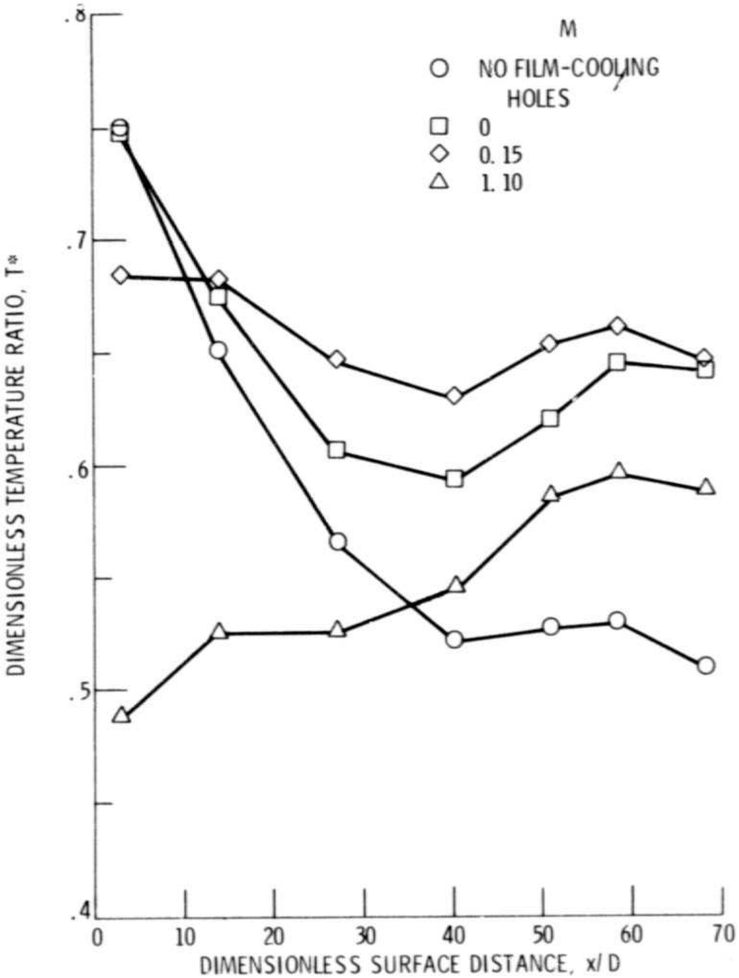


(a) Cross-sectional midspan view showing internal cooling scheme and thermocouple locations are shown as a composite of vane 2 and vane 3.



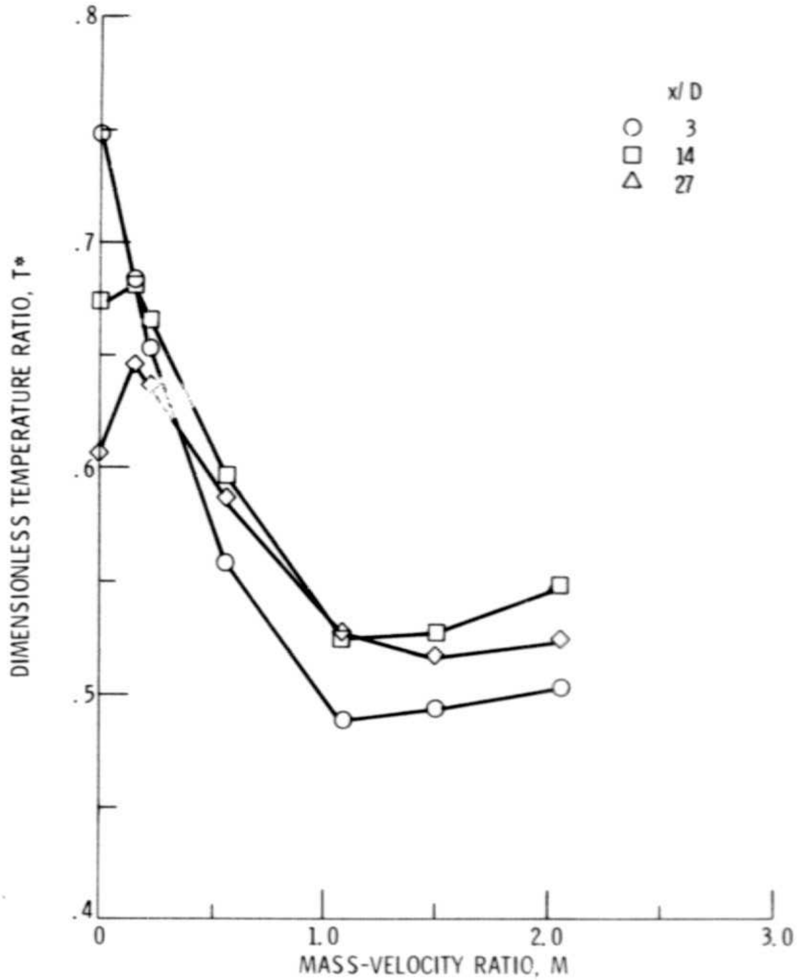
(b) Cutaway view.

Figure 2. - Schematic view of J-75 size test vanes.



(a) EFFECTS OF BLOWING IN A NON-TURBULENT BOUNDARY LAYER.

Figure 3. - Dimensionless temperature distributions on the suction surface of a turbine vane. Mid-chord convection flow, 1.85 percent.



(b) SELECTED WALL TEMPERATURES VERSUS MASS-VELOCITY RATIO.

Figure 3. - Concluded.

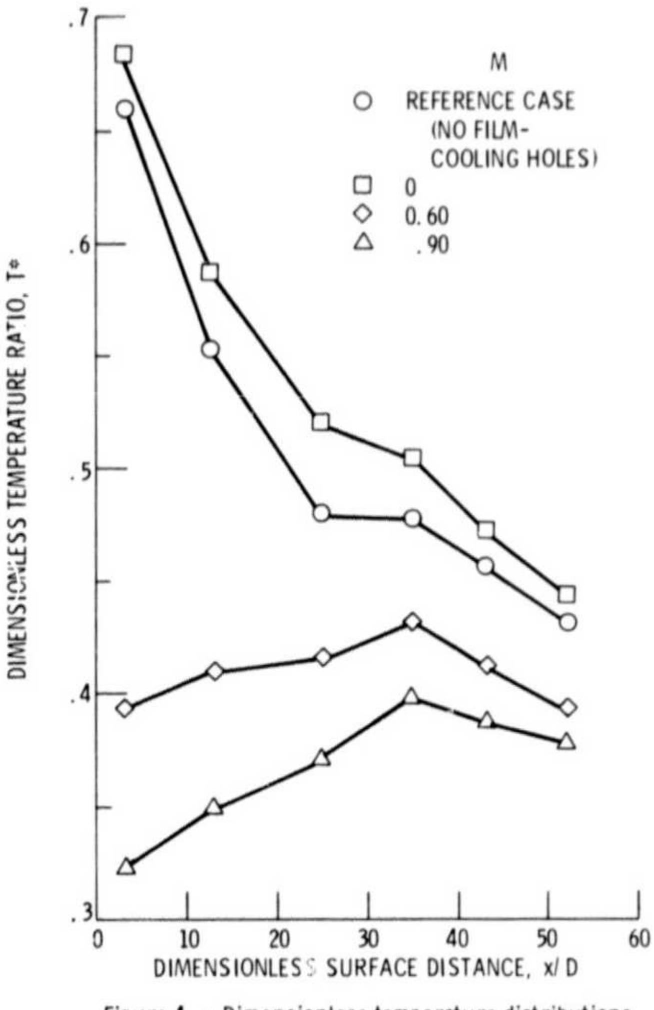


Figure 4. - Dimensionless temperature distributions on the pressure surface of a turbine vane. Mid-chord coolant flow, 3.8 percent.

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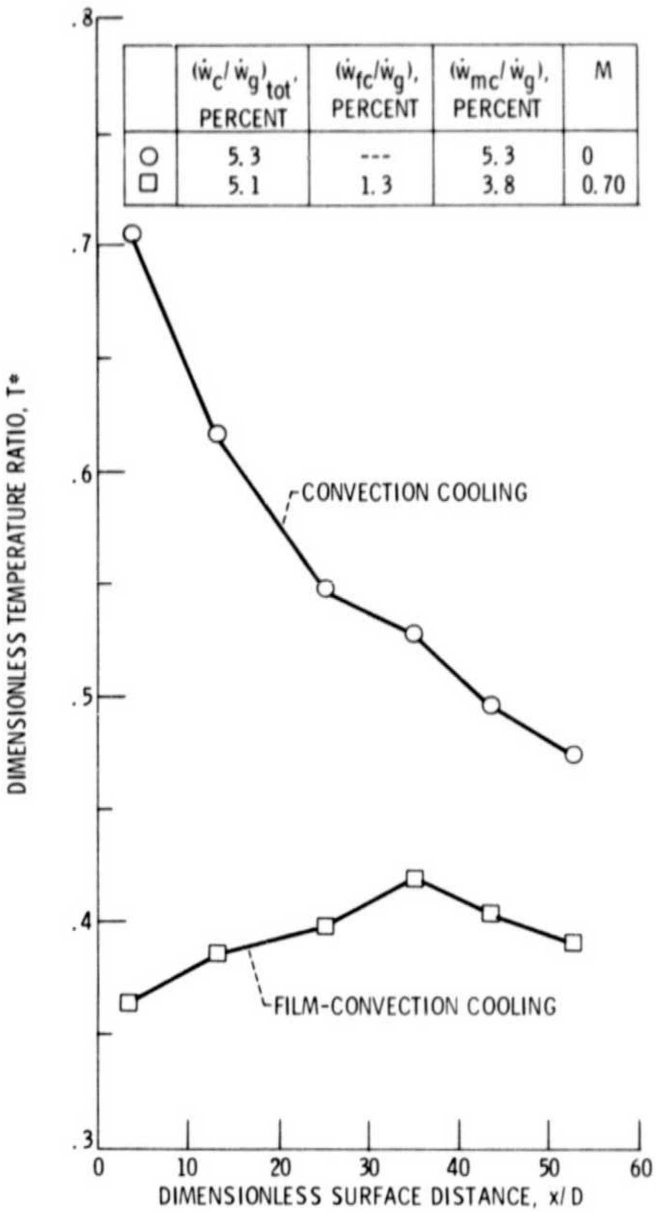
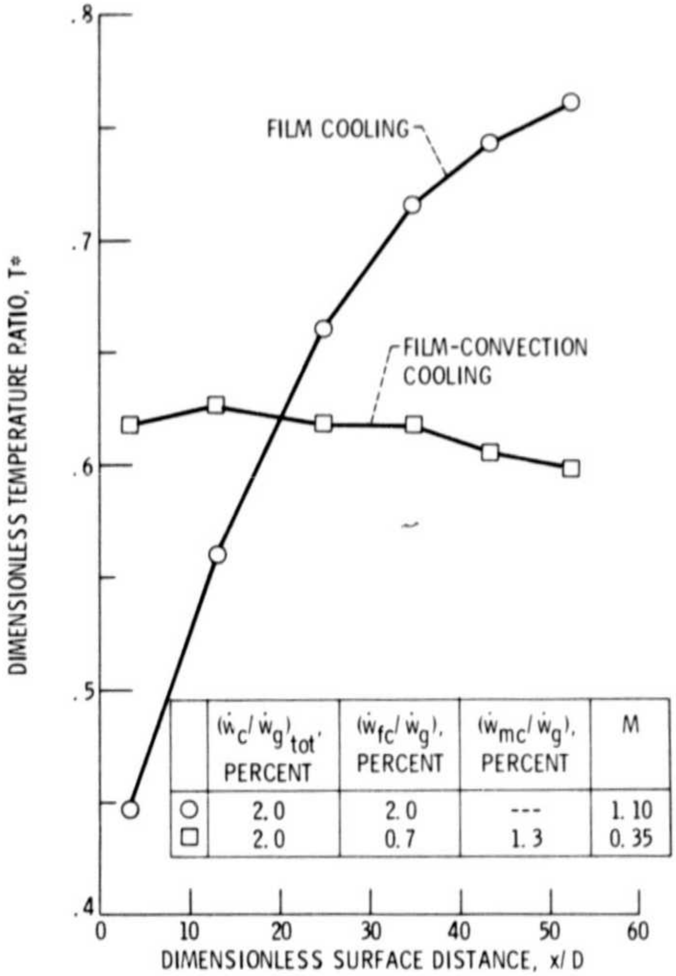


(a) FILM-CONVECTION VERSUS CONVECTION COOLING.

Figure 5. - Dimensionless temperature distributions showing the relative benefits of combining film and convection cooling in a turbulent boundary layer (vane pressure surface).



(b) FILM-CONVECTION VERSUS FILM COOLING.

Figure 5. - Concluded.